PATENT SPECIFICATION

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PROVISIONAL SPECIFICATION

Improvements in Speed Governors for Variable Pitch Screw-Propellers especially for Aircraft

I, KENNETH VICTOR DIPROSE, a British Subject, of the Royal Aircraft Establishment, South Farnborough, Hampshire, do hereby declare the nature of this invention to be as follows:—

This invention relates to speed governors for variable pitch screw-propellers, especially for aircraft, and refers more particularly to improvements in speed governors of the type exemplified by the conventional "constant speed unit" commonly used in aircraft and comprising a system of centrifugal flyweights balanced against a spring (whose abutment may be displaceable, e.g. by a rack and pinion mechanism, to vary the equilibrium speed of the flyweight system), and controlling the displacement of a movable control member, e.g. a hydraulic valve piston, in such a way that its displacement from a neutral position in either direction is substantially proportional to the speed error (positive or negative). The control member operates to control the application of power, e.g. hydraulic or electric, to the pitch changing mechanism of the propeller and ideally the arrangement is such that the pitch is changed, in the appropriate sense for correcting the speed error, at a rate proportional to the displacement of the control member from its neutral position and hence proportional to the speed error.

The dynamics of such a system, having regard to the aerodynamic characteristics of the propeller and the torque-speed characteristics of the engine driving it, are such that the speed error ideally obeys a linear differential equation of the second order, whose solution for "free motion", i.e. that following a transient disturbance, such as a sudden increase of engine power, a sudden change of "selected" speed or a sudden change of advance per revolution due to a change of forward speed, is of the form of a subsidence or an oscillation according to the amount of damping present. When rate of pitch change is proportional to speed error, the damping is supplied solely by the propeller itself, and is proportional to the difference of the propeller (braking) and engine torque/speed gradients. This damping may be called aerodynamic damping; and in general it is insufficient to make the control dead-beat, the motion having the character of a damped oscillation. Damping decreases with increase of "sensitivity", i.e. increase of the ratio, pitch change rate/speed error, which ratio is preferably large in order to reduce both the initial speed error surge occurring on a change of engine torque due to manipulation of the "throttle", and the steady state speed error arising from rapid acceleration in the direction of travel. With increase of altitude or decrease of forward speed the damping also decreases. Further, time lag in the governor system, which is always present to a greater or less extent, decreases the effective damping and in some installations (depending on the engine characteristics) the effective damping may become zero or negative at great altitudes or/and small forward speeds ("take-off" case), so that the oscillation becomes unstable in such circumstances.

The broad object of the invention is therefore to increase the damping of the system and in this connection it is pointed out that an increase of damping will not only cause the oscillation (of speed error) to die away more quickly (and if sufficiently great will cause the motion to be dead-beat), but will greatly reduce the initial speed error surge after a disturbance of equilibrium, such as a sudden increase of engine torque. This object is attained by providing artificial or "control" damping, by so constructing the governor that (ideally) the rate of pitch change is (independently and additively) proportional not only to the (instantaneous) value of the speed error but also to the rate at which the speed error increases.

Broadly, the invention consists in

[*Price* 1/-]

combining with a centrifugal governor system of the type first herein referred to, a rotary inertia system comprising a mass rotated by the driving shaft of the centrifugal governor, and a spring connection resisting angular displacement of the mass relatively to the shaft, the load in the last named spring being transmitted to the control member (valve piston or the like), so that the displacement of the latter from its neutral position is proportional to the sum of two forces, of which one, derived from the centrifugal system, is proportional to the (instantaneous) speed error and the other, derived from the rotary inertia system, is proportional to the (instantaneous) rate of increase of the speed error.

This may be accomplished in various ways. According to one feature of the invention the spring used for balancing the centrifugal force of the fly-weight system is also utilized for resisting the displacement of the rotory mass.

According to another feature of the invention, the whole of the flyweight system including the flyweights themselves and their carrier (which may include the conventional oil shield) is utilised as the rotary mass.

According to yet another feature of the invention the load in the spring resisting angular displacement of the rotary mass is transmitted to the control member by a force-intensifying hydraulic relay, which multiplies the spring load by a substantially constant factor.

In order that the nature of the invention may be fully understood three typical examples are hereinafter briefly described with reference to the accompanying diagrammatic drawings, of which:

Fig. 1 shows in perspective the essential elements of a first example,

Fig. 2 illustrates in a similar manner a second example,

Fig. 3 shows the essential elements of a third example in central axial section, and

Figs. 4 and 5 are detail sectional views taken along the lines 4—4 and 5—5 respectively of Fig. 3.

In Fig. 1 the stem of the control member, e.g. a hydraulic valve piston is shown at 10; this can slide axially and rotates with the flyweight assembly, of which one of the flyweights is indicated at 15, being pivoted at 16 on the weight carrier (not shown) and having a short lever 17 which bears on the head 18 of the stem 10. The centrifugal force of the flyweights 15 is opposed by a spring 14 bearing on the underside of head 18. The abutment of spring 14 is axially displaceable by means of a rack 11, sector 12, and speed selector lever 13. As the head 18 rotates and the spring 14 does not, in practice a ball race would be inserted between these two members, but this is omitted in the drawing for clearness.

In addition to this centrifugal controlling mechanism a second system of flyweights is provided, one of which is shown at 19 pivoted to the weight carrier about an axis 20 which intersects the axis of control member 10. Flyweight 19 has a short lever 21 which also bears on the head 18 of the stem 10.

The direction of rotation is shown by an arrow, and it will be seen that if the speed of rotation of the system increases the flyweight 19 will tend to lag and be displaced about its pivot 20, so that the lever 21 will exert pressure on the head 18 in opposition to the spring 14. The force exerted by it is therefore balanced by the sum of the forces exerted by levers 17 and 21, so that the displacement of the stem 10 from the equilibrium position is proportional to the sum of the force exerted by lever 21, which is proportional to the angular acceleration of the weight carrier and the excess of the force exerted by lever 17 over that required to maintain the stem 10 in the equilibrium position, which excess is approximately proportional to the speed error, so long as the latter is small relatively to the absolute speed.

As the drawings are only intended to illustrate diagrammatically the principle of the apparatus, for convenience the flyweight 19 is shown as only having one lever 21 which exerts a force on the head 18 when the system is accelerated. In practice, the lever 21 could be made to engage a groove in the head 18, so that it would be operated in both directions.

In the example illustrated in Fig. 2, the stem 10 of the valve, or other controlling member, is arranged coaxially within a driving sleeve 22 carrying a driving pinion 23, with respect to which the stem 10 is free to slide axially and rotate. As before, the head 18 of stem 10 is engaged on the one side by levers 17 carried by the centrifugal flyweights 15 and on the other side by the spring 14; the adjustable abutment by which the equilibrium speed is selected being in this case omitted for clearness. As before, the flyweights 15 are pivoted at 16 on a carrier, which in this example takes the form of an oil shield 24 free to rotate on the driving sleeve 22. The latter carries a two armed lever 25 connected by links 27 with the flyweights 15. One end of each link 27 is pivoted at 26 to the

lever 25 and the other end of the link is connected to the flyweight 15 by a pivot 28 which is offset from the fulcrum pivot 16 of the flyweight.

The drive is therefore transmitted from the driving sleeve 22 to the carrier 24 and flyweights 15 through the lever 25 and links 27. As before, the direction of rotation is shown by an arrow, and it will be seen that when the system is accelerated, the accelerating force, transmitted by links 27 to the flyweights 15, exerts a moment about each fulcrum pivot 16 tending to throw the flywheight outwards and these moments are additive to the centrifugal moments of the flyweights 15. Similarly, when the system is decelerated, the decelerating forces transmitted by links 27 exert moments about the pivots 16 opposed to the centrifugal moments of the flyweights 15. The force applied to head 18 by the levers 17 is therefore the algebraic sum of a force proportional to the centrifugal force exerted by the flyweights 15 and a force proportional to the moment required to accelerate (or decelerate), the whole flyweight system comprising the flyweights themselves and their carrier 24; and the displacement from its equilibrium position of the control member 10, 18 is therefore proportional to the alegbraic sum of the " acceleration " component of the force exerted by the levers 17 and the excess (or defect) of the " centrifugal " component of the force exerted by levers 17 over that exerted in the equilibrium condition. The excess centrifugal force component acting on the head 18 is approximately proportional to the speed error, provided the latter is small, and the " acceleration " component of the total force is proportional to the acceleration.

Since the magnitude of the " acceleration " forces transmitted to the valve or like control member are proportional to the inertia of the masses whose acceleration is utilised to effect displacement of the control member, it will be seen that the displacement of the control member corresponding to a given acceleration, and hence the amount of " artificial " or " control " damping of the propeller-governor system that can be so provided, can be made much greater in the example illustrated in Fig. 2 than in the example illustrated in Fig. 1, because in the arrangement of Fig. 2 the inertia of the whole of the flyweight system including the combined oil shield and carrier 24 is utilised in this way, whereas in the arrangement of Fig. 1 only the inertia of the flyweights 19 is so used and practical considerations severely limit the size of the flyweights 19 that can be accommodated.

More powerful damping of the propeller governor system than is possible with the arrangement of Fig. 2 can be obtained by the incorporation of hydraulic relay or intensifier as in the typical example illustrated in Fig. 3. In this example the invention is applied to the governor of a hydraulic variable pitch propeller and the arrangement illustrated is appropriate to an installation operating on the " single line " system, in which high pressure oil is utilised for coarsening the pitch only, the change of pitch in the reserve direction being effected by engine oil pressure assisted by the centrifugal twisting moment of the propeller blades.

Referring to Fig. 3, the control member 10 is the stem of a piston valve having lands 29, 30 co-operating with ports 31, 32, 33, formed in the driving sleeve 22. Port 31 communicates with a high pressure feed line 34 and port 32 with drain, port 33 being in communication with a delivery line 35 by which high pressure oil is led to the variable pitch mechanism of the propeller for coarsening the pitch. As in the examples of Figs. 1 and 2, the valve stem 10 terminates in a head 18 engaged by levers 17 of flyweights 15 pivoted at 16 on the combined weight carrier and oil shield 24. As before the force exerted by the flyweight levers 17 is opposed by that exerted on the opposite face of the head 18 by a control spring which is not actually shown in Fig. 3 but is similar to the spring 14 of Figs. 1 and 2. In Fig. 3 the valve stem is shown in the equilibrium position; a " plus " speed error will cause the flyweights 15 to swing outwards and raise the valve stem 10, thus connecting the propeller line 35 to the pressure line 34 so that pressure is applied to the pitch changing mechanism to coarsen the pitch. Similarly, a " minus " speed error will cause the valve stem to descend, thus connecting the propeller line 35 to drain, through port 32, and allowing the pitch of the propeller to fine off under the action of the centrifugal twisting moment and engine oil pressure. The ports 31, 32 are so proportioned that the rate at which the oil can pass into or out of the propeller line 35, and hence the rate of pitch change, is approximately proportional to the displacement of the valve stem 10 from its epuilibrium position in either direction.

As in Fig. 2 the flyweight carrier 24 can rotate on the driving sleeve 22 and in this instance the drive is transmitted

from the sleeve 22 to the carrier 24 by a coil spring 47. If the system is accelerated or decelerated the moment required to accelerate or decelerate the carrier 24 and the flyweight system carried thereby, which moment is equal to the rotary inertia of the carrier 24 and the masses supported thereon multiplied by the angular acceleration, is transmitted by the spring 47; and the angular lag or lead established between the carrier 24 and the driving sleeve 22 is equal to the accelerating moment as above defined divided by the elasticity of the spring 47. The angular lag or lead thus established is therefore proportional to the acceleration of the rotary mass 24, etc., multiplied by its inertia.

The angular lag or lead of the weight carrier 24 is translated into a proportional displacement of the valve stem 10 in the appropriate direction, additional to its displacement due to the action of the centrifugal flyweight system, by means of a hydraulic relay comprising an enlargement 37 of the valve stem 10, which constitutes a piston sliding in a double acting cylinder formed by an enlargement 36 in the bore of the driving sleeve 22. The upper and lower spaces 36*a* and 36*b* of cylinder 36 communicate respectively by means of passages 41, 42, formed in the driving sleeve 22, with a port 40 formed in a sleeve extension 39 of the flyweight carrier 24, which port in turn communicates with a branch lead 38 from the high pressure lead 34. The driving sleeve 22 also has ports 43, 44 communicating respectively with the cylinder spaces 36*a*, 36*b* and aligned in the axial direction (as seen in Fig. 3) with ports 45, 46 formed in the sleeve extension 39 of the carrier 24; and both the latter ports communicate with drain.

Ports 45, 46 are offset from one another in the circumferential direction and are so located that when the system is unaccelerated their overlaps with the respectively associated ports 43, 44 in the driving sleeve 22 are equal as shown in Figs. 4 and 5 with the result that the pressures in the cylinder spaces 36*a*, 36*b* on either side of the piston 37 are equal. when, however, the sleeve extension 39 is angularly displaced with respect to the driving sleeve 22 owing to acceleration or deceleration of the system, the overlaps of ports 43, 45 and 44, 46 respectively become unequal and a pressure difference is therefore established across the piston 37 proportional to the angular displacement of the sleeve 39, 22. It will be seen from Figs. 4 and 5, in which the direction of rotation is indicated by arrows, that when the sleeve 39 lags corresponding to an acceleration of the system, the overlap of ports 43, 45 is increased and that of ports 44, 46 is decreased, with the result that pressure in space 36*b* below the piston 37 exceeds that in space 36*a* above it. The unbalanced force on the piston 37 tends to raise the valve stem 10 and thereby place the propeller line 35 in communication with the high pressure line 34 for coarsening the propeller pitch. Similarly, if the system is decelerated a pressure difference in the opposite direction is established across the piston 37 tending to depress the valve stem and place the propeller line 35 in communication with drain through the port 32.

Since the valve stem is subject to the load of the control spring (not shown) its displacement in either direction, due to acceleration or deceleration of the system, is approximately proportional to the pressure difference established across the piston 37; and this in turn is approximately proportional to the difference between the overlaps of ports 43, 45 and 44, 46 respectively, and hence to the magnitude of the acceleration or deceleration, as hereinbefore explained. In order to provide the hydraulic relay system 36—46 with adequate damping, not to be confused with damping of the whole propeller/governor system, and prevent hunting of the hydraulic relay, it is preferable that the ports 43, 45 and 44, 46 respectively should have a positive overlap in the neutral position as shown in Figs. 4 and 5. Such an overlap is necessarily accompanied by a continuous leakage of oil through the hydraulic relay and if this leakage were large the consequential pressure drop in the propeller line 35 would be detrimental; but the overlap of ports 43, etc. as above can be made sufficiently small for the leakage and the accompanying pressure drop in the propeller line to be kept within an acceptably low limit.

It will be seen that the magnitude of the force applied to the valve stem 10 in response to accelerations and decelerations of the system is dependent on the supply pressure in line 34 and the effective area of the piston 37 and not on the inertia of the rotary mass 24, etc.; and it can be shown that with operating pressures of the order usual in conventional constant speed units for aircraft variable pitch propellers and using a piston 37 of an area that can easily be accommodated very much larger valve displacing forces can be developed in response to acceleration than in any system in which the valve displacing forces are derived directly from the inertia of the rotary masses, as exemplified in Figs. 1 and 2.

The hydraulic relay system shown in Fig. 3 therefore acts as a valve-operating force-intensifier and enables much more powerful "control" damping of the whole propeller-governor system to be provided than with any system relying on the mechanical application of forces due to the inertia of such rotary masses as can conveniently be accommodated.

Dated this 25th day of July, 1945.

C. STRATTON CROSS,
Agent for the Applicant.

COMPLETE SPECIFICATION

Improvements in Speed Governors for Variable Pitch Screw-Propellers especially for Aircraft

I, KENNETH VICTOR DIPROSE, a British Subject, of the Royal Aircraft Establishment, South Farnborough, Hampshire, do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

This invention relates to speed governors for variable pitch screw-propellers, especially for aircraft, and refers more particularly to improvements in speed governors of the type exemplified by the conventional "constant speed unit" commonly used in aircraft and comprising a system of centrifugal flyweights balanced against a spring (whose abutment may be displaceable, e.g., by a rack and pinion mechanism, to vary the equilibrium speed of the flyweight system), and controlling the displacement of a movable control member, e.g., a hydraulic valve piston, in such a way that its displacement from a neutral position in either direction is substantially proportional to the speed error (positive or negative). The control member operates to control the application of power, e.g., hydraulic or electric, to the pitch changing mechanism of the propeller and ideally the arrangement is such that the pitch is changed, in the appropriate sense for correcting the speed error, at a rate proportional to the displacement of the control member from its neutral position and hence proportional to the speed error.

The dynamics of such a system, having regard to the aerodynamic characteristics of the propeller and the torque/speed characteristics of the engine driving it, are such that the speed error ideally obeys a linear differential equation of the second order, whose solution for "free motion", i.e., that following a transient disturbance, such as a sudden increase of engine power, a sudden change of "selected" speed or a sudden change of advance per revolution due to a change of forward speed, is of the form of a subsidence or an oscillation according to the amount of damping present. When rate of pitch change is proportional to speed error, the damping is supplied solely by the propeller itself, and is proportional to the difference of the propeller (braking) and engine torque/speed gradients. This damping may be called aerodynamic damping; and in general it is insufficient to make the control deadbeat, the motion having the character of a damped oscillation. Damping decreases with increase of "sensitivity", i.e., increase of the ratio, pitch change rate/speed error, which ratio is preferably large in order to reduce both the initial speed error surge occurring on a change of engine torque due to manipulation of the "throttle", and the steady state speed error arising from rapid acceleration in the direction of travel. With increase of altitude or decrease of forward speed the damping also decreases. Further, time lag in the governor system, which is always present to a greater or less extent, decreases the effective damping and in some installations (depending on the engine characteristics) the effective damping may become zero or negative at great altitudes or/and small forward speeds ("take-off" case), so that the oscillation becomes unstable in such circumstances. An increase of damping will not only cause the oscillation (of speed error) to die away more quickly (and if sufficiently great will cause the motion to be dead-beat), but will greatly reduce the initial speed error surge after a disturbance of equilibrium, such as a sudden increase of engine torque. The invention is concerned with speed governors in which such increased damping is obtained by the provision of artificial or "control" damping that (ideally) the rate of pitch change is (independently and additively) proportional not only to the (instantaneous) value of the speed error but also to the rate at which the speed error increases.

To be really effective, a control in accordance with rate of change of speed error must be able to impose forces of considerable magnitude, ideally of the

order of those imposed by a centrifugal system operating in accordance with instantaneous values of propeller speed: the present invention is based on a realisation of this fact and provides speed governors by means of which satisfactory control may be achieved without incurring excessive penalties of weight and space.

In speed governors in accordance with the invention, a movable pitch-control member receives control forces from a centrifugal flyweight system and from a rotary inertia system, the forces received from the rotary inertia system being applied through a force-intensifying relay which subjects them to an amplification to which the forces received from the centrifugal system are not subject. The moment of inertia of the rotary inertia system is as large as conveniently possible and it is preferred to make use of as much as possible of the essential rotating mass, such as the whole of the flyweight system with its flyweights and their carrier (which may include the conventional oil shield).

Other features of the invention are incorporated in the typical example of mechanism in accordance with it which will now be described with reference to the accompanying diagrammatic drawings:—

Fig. 1 shows the essential elements in central axial section, and

Figs. 2 and 3 are detail sectional views taken along the lines II—II and III—III, respectively, of Fig. 1.

In the example illustrated, the invention is applied to the governor of a hydraulic variable pitch propeller and the arrangement is appropriate to an installation operating on the "single line" system, in which high pressure oil is utilised for coarsening the pitch only, the change of pitch in the reverse direction being effected by engine oil pressure assisted by the centrifugal twisting moment of the propeller blades.

As shown, the stem 10 of a piston valve, having lands 29 and 30, terminates in a head 18 engaged by levers 17 of flyweights 15 pivoted at 16 on the combined weight carrier and oil shield 24. The force exerted by the flyweight levers 17 is opposed by that exerted on the opposite face of the head 18 by a control spring 14. The initial compression of the spring 14 is variable by means of an adjustable abutment (not shown). The lands 29 and 30 of the piston valve co-operate with ports 31, 32, 33, formed in the driving sleeve 22. Port 31 communicates with a high pressure feed line 34 and port 32 with drain, port 33 being in communication with a delivery line 35 by which high pressure oil is led to the variable pitch mechanism of the propeller for coarsening the pitch. In Fig. 1 the valve stem is shown in the equilibrium position; a "plus" speed error will cause the flyweights 15 to swing outwards and raise the valve stem 10, thus connecting the propeller line 35 to the pressure line 34 so that pressure is applied to the pitch changing mechanism to coarsen the pitch. Similarly, a "minus" speed error will cause the valve stem to descend, thus connecting the propeller line 35 to drain, through port 32, and allowing the pitch of the propeller to fine off under the action of the centrifugal twisting moment and engine oil pressure. The ports 31, 32 are so proportioned that the rate at which the oil can pass into or out of the propeller line 35, and hence the rate of pitch change, is approximately proportional to the displacement of the valve stem 10 from its equilibrium position in either direction.

The flyweight carrier 24 can rotate on the driving sleeve 22 and the drive is transmitted from the sleeve 22 to the carrier 24 by a coil spring 47. If the system is accelerated or decelerated the moment required to accelerate or decelerate the carrier 24 and the flyweight system carried thereby, which moment is equal to the moment of inertia of the carrier 24 and the masses supported thereon multiplied by the angular acceleration, is transmitted by the spring 47; the angular lag or lead established between the carrier 24 and the driving sleeve 22 is equal to the accelerating moment as above defined divided by the elasticity of the spring 47. The angular lag or lead thus established is therefore proportional to the acceleration of the rotary mass 24, etc., multiplied by its inertia.

The angular lag or lead of the weight carrier 24 is translated into a proportional displacement of the valve stem 10 in the appropriate direction, additional to its displacement due to the action of the centrifugal flyweight system, by means of a hydraulic relay comprising an enlargement 37 of the valve stem 10, which constitutes a piston sliding in a double acting cylinder formed by an enlargement 36 in the bore of the driving sleeve 22. The upper and lower spaces 36a and 36b of cylinder 36 communicate respectively by means of passages 41, 42, formed in the driving sleeve 22, with a port 40 formed in a sleeve extension 39 of the flyweight carrier 24, which port in turn communicates with a branch lead 38

from the high pressure lead 34. The driving sleeve 22 also has ports 43, 44 communicating respectively with the cylinder spaces 36*a*, 36*b*, and aligned in the axial direction (as seen in Fig. 1) with ports 45, 46 formed in the sleeve extension 39 of the carrier 24; and both the latter ports communicate with drain.

Ports 45, 46 are offset from one another in the circumferential direction and are so located that when the system is unaccelerated their overlaps with the respectively associated ports 43, 44 in the driving sleeve 22 are equal as shown in Figs. 2 and 3 with the result that the pressures in the cylinder spaces 36*a*, 36*b* on either side of the piston 37 are equal. When, however, the sleeve extension 39 is angularly displaced with respect to the driving sleeve 22 owing to acceleration or deceleration of the system, the overlaps of ports 43, 45 and 44, 46 respectively become unequal and a pressure difference is therefore established across the piston 37 proportional to the angular displacement of the sleeve 39, 22. It will be seen from Figs. 2 and 3, in which the direction of rotation is indicated by arrows, that when the sleeve 39 lags corresponding to an acceleration of the system, the overlap of ports 43, 45 is increased and that of ports 44, 46 is decreased, with the result that pressure in space 36*b* below the piston 37 exceeds that in space 36*a* above it. The unbalanced force on the piston 37 tends to raise the valve stem 10 and thereby place the propeller line 35 in communication with the high pressure line 34 for coarsening the propeller pitch. Similarly, if the system is decelerated a pressure difference in the opposite direction is established across the piston 37 tending to depress the valve stem and place the propeller line 35 in communication with drain through the port 32.

Since the valve stem is subject to the load of the control spring 14 its displacement in either direction, due to acceleration or deceleration of the system, is approximately proportional to the pressure difference established across the piston 37; and this in turn is approximately proportional to the difference between the overlaps of ports 43, 45 and 44, 46 respectively, and hence to the magnitude of the acceleration or deceleration, as hereinbefore explained. In order to provide the hydraulic relay system 36—46 with adequate damping, not to be confused with damping of the whole propeller-governor system, and prevent hunting of the hydraulic relay, it is preferable that the ports 43, 45, and 44, 46 respectively should have a positive overlap in the neutral position as shown in Figs. 2 and 3. Such an overlap is necessarily accompanied by a continuous leakage of oil through the hydraulic relay and if this leakage were large the consequential pressure drop in the propeller line 35 would be detrimental; but the overlap of ports 43, etc., as above can be made sufficiently small for the leakage and the accompanying pressure drop in the propeller line to be kept within an acceptably low limit.

The total control force applied to the control member 10, 18, will be the algebraic sum of a force proportional to the centrifugal force exerted by the flyweights 15 and a force proportional to the moment required to accelerate (decelerate) the whole flyweight system: the instantaneous displacement from its equilibrium position of the control member is proportional to the algebraic sum of the " acceleration " force (exerted by the hydraulic relay) and the excess (or defect) of the " centrifugal " force (exerted by the flyweights) over that required to balance the spring 14 in the equilibrium.

The hydraulic relay system shown acts as a valve-operating force-intensifier and it will be seen that the magnitude of the force applied to the valve stem 10 in response to accelerations and decelerations of the system is dependent on the supply pressure in line 34 and the effective area of the piston 37 and not only on the inertia of the rotary mass 24 etc.; it can be shown that with operating pressures of the order usual in conventional constant speed units for aircraft variable pitch propellers and using a piston 37 of an area that can easily be accommodated very much larger valve displacing forces can be developed in response to acceleration than in any system in which the valve displacing forces are derived directly from the inertia of rotary masses of practicable magnitude. The utilisation of the whole of the rotary masses of the flyweight system enables adequately powerful " control " damping of the whole propeller-governor system to be obtained.

Having now particularly described and ascertained the nature of my said invention and in what manner the same is to be performed, I declare that what I claim is:—

1. A speed governor for a variable pitch screw propeller comprising a movable pitch-control member arranged to receive operating forces from a centrifugal flyweight system sensitive to changes in propeller speed and from a rotary inertia system sensitive to rate of change of pro-

peller speed, the forces received from the rotary inertia system being applied through a force-intensifying relay which subjects them to an amplification to which the forces received from the centrifugal system are not subject.

2. A speed governor as claimed in claim 1 in which the rotary inertia system comprises the whole of the rotating mass of the centrifugal flyweight system.

3. A speed governor as claimed in either of the preceding claims in which the rotary inertia system is driven by way of a flexible coupling and the angular lag or lead of the inertia system in relation to a member by which drive is imparted to the coupling determines the magnitude of the operating force which is to be imparted by the relay.

4. A speed governor as claimed in any of the preceding claims in which the rotary inertia system operates through a hydraulic relay acting upon the movable pitch-control member by means of differential pressures.

5. A speed governor as claimed in claim 4 and in which the hydraulic relay comprises members operatively coupled to the rotary inertia system and a driving member respectively and co-operating to control the hydraulic pressures acting on the opposite sides of a piston integral with or connected to the movable pitch-control member.

6. A speed governor as claimed in claim 5 in which the members operatively coupled to the inertia system and to the driving member respectively have formed in them passages which co-operate to control the bleed to drain of hydraulic fluid supplied under pressure to the opposite sides of the piston.

7. A speed governor as claimed in any of the preceding claims in which the movable pitch-control member is an axially movable valve member which controls the flow of liquid in a hydraulic pitch-changing system and is operatively connected to a piston forming part of the relay and sliding in a cylinder which is arranged for rotation about its axis and has at each end a port connected to a supply of pressure fluid and a port leading to drain, the bleed of fluid to drain being controlled by co-operating ports in a part of the centrifugal flyweight system mounted coaxially with the cylinder and driven therewith through a coil spring.

8. A speed governor for a variable pitch screw propeller substantially as described with reference to the accompanying drawings.

Dated this 10th day of July, 1946.
C. STRATTON CROSS,
Chartered Patent Agent,
Agent for the Applicant.

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[This Drawing is a reproduction of the Original on a reduced scale.]

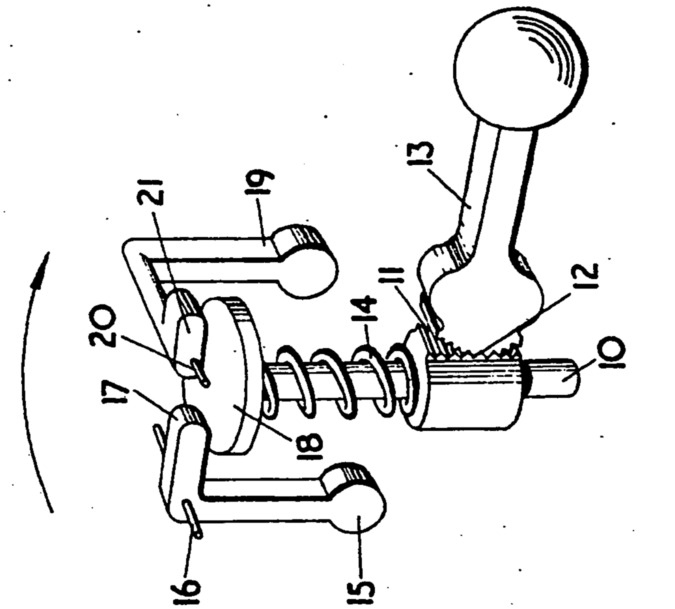


Fig. 1.

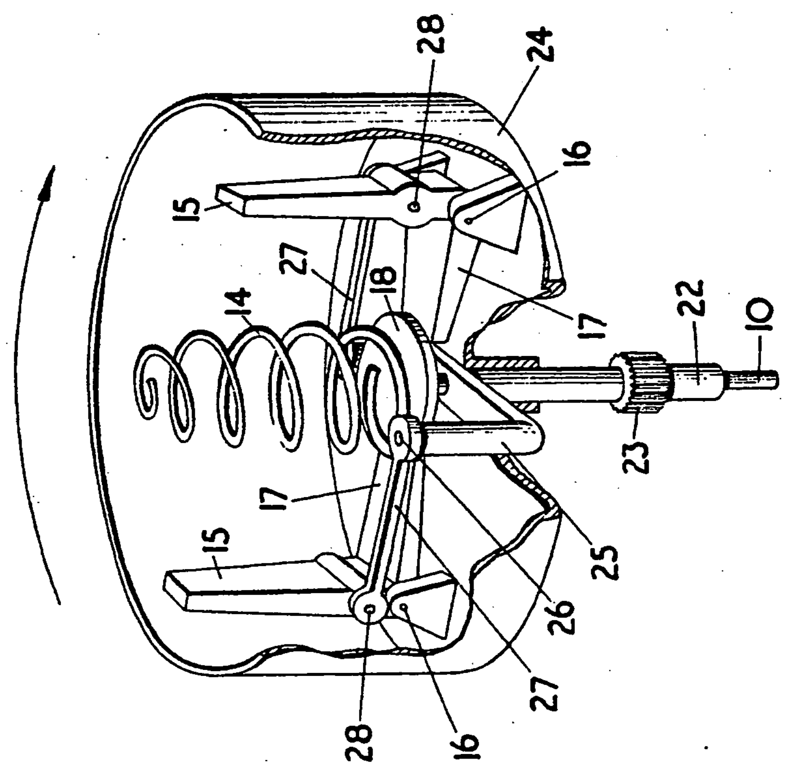


Fig. 2.

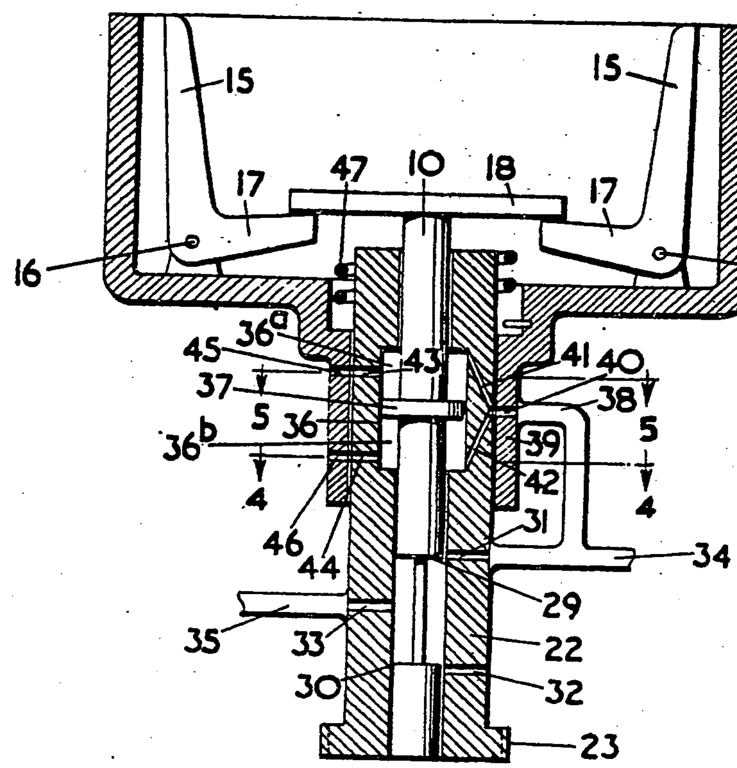


Fig. 3.

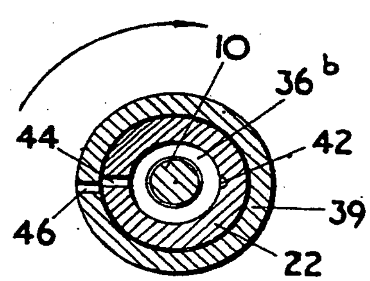


Fig. 4.

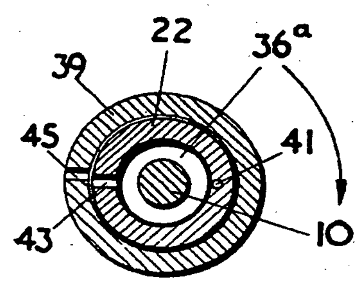


Fig. 5.

[This Drawing is a reproduction of the Original on a reduced scale.]

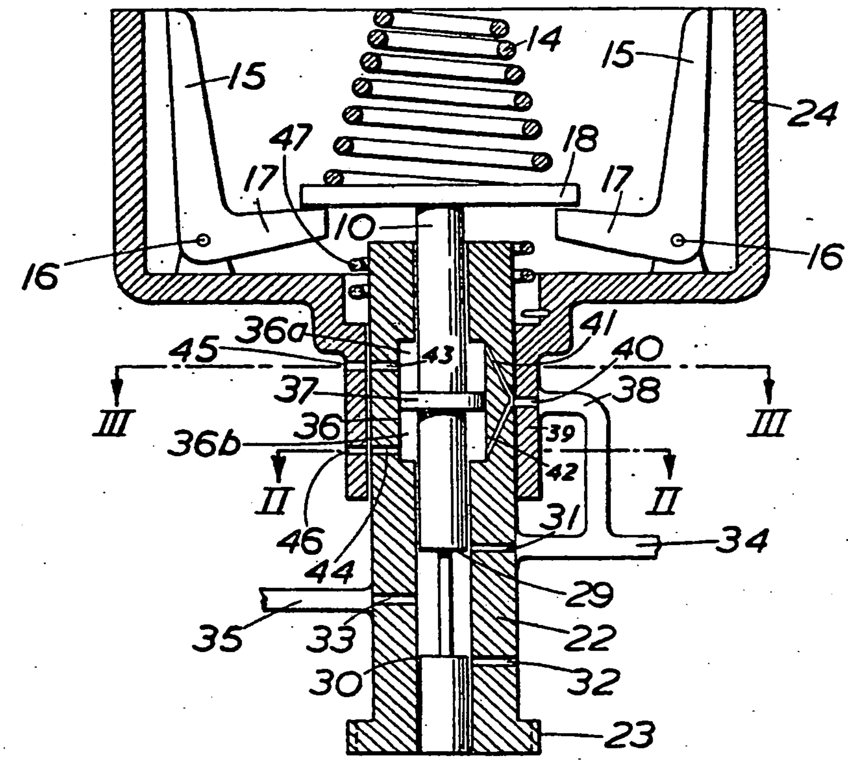
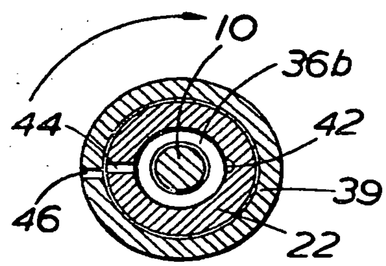


Fig. 1



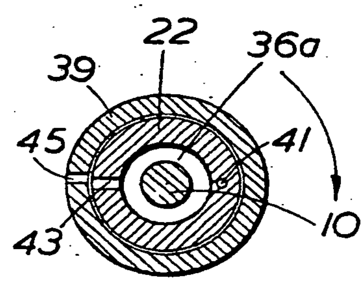


Fig. 2

Fig. 3

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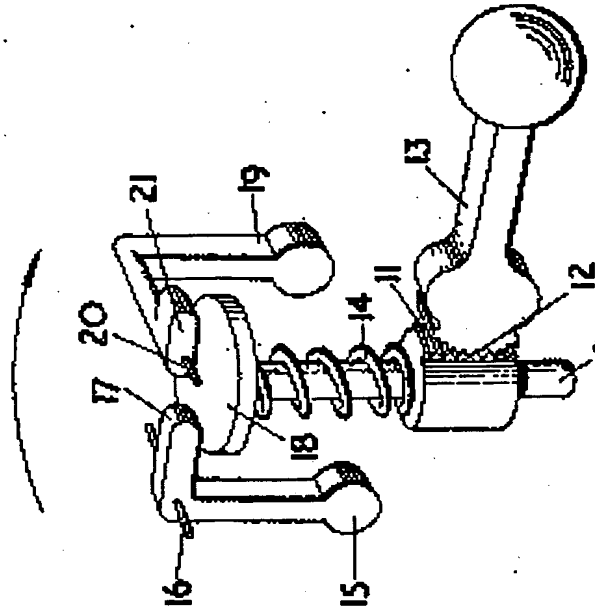

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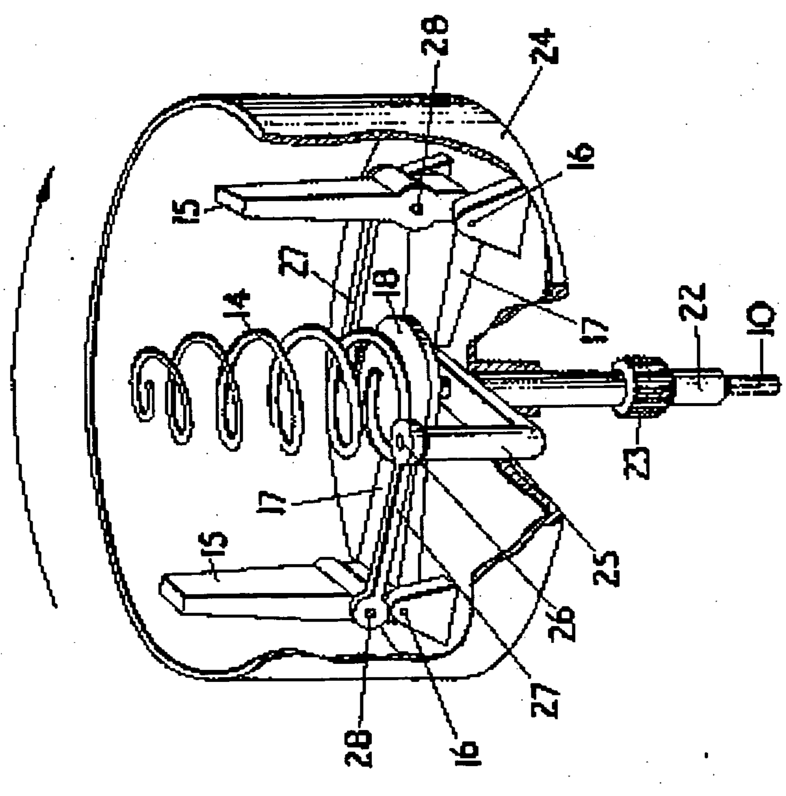

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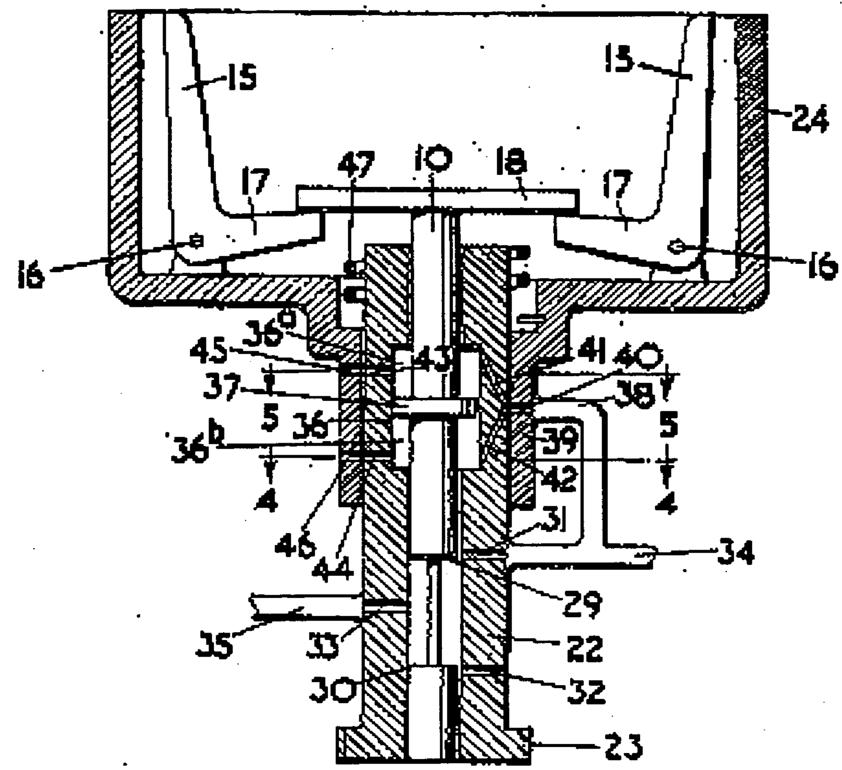

Fig. 3.

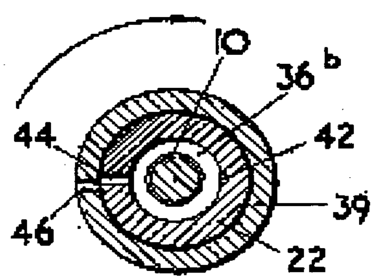

Fig. 4.

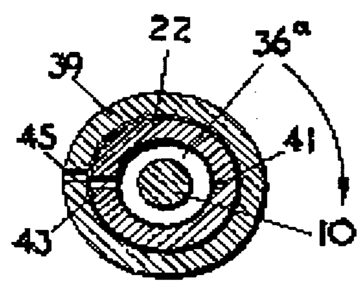

Fig. 5.

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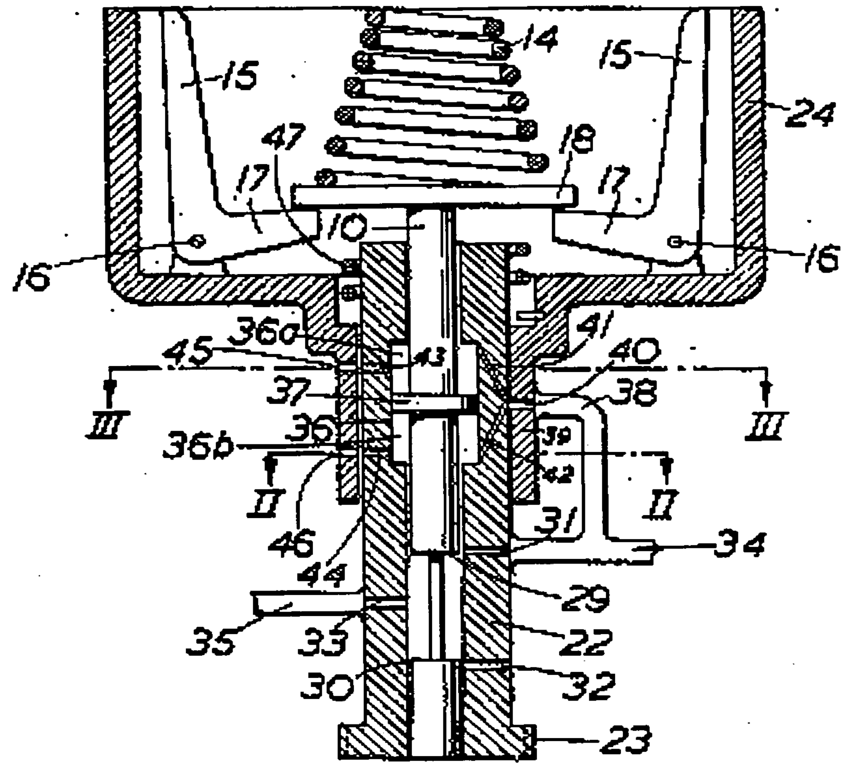

Fig. 1

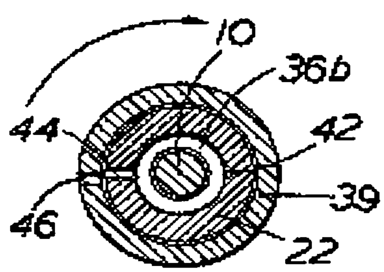

Fig. 2

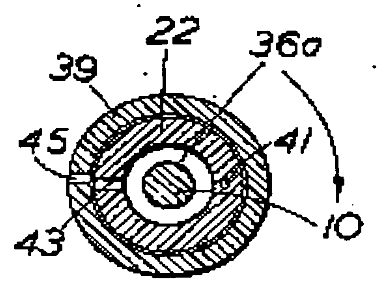

Fig. 3